CAPACITY CONTROL VALVE
FOR VARIABLE DISPLACEMENT COMPRESSOR

Cross-References to Related Applications, If Any:

This application claims priority of Japanese Application No.2002-278764 filed on September 25, 2002 and entitled "Capacity Control Valve for Variable Displacement Compressor".

BACKGROUND OF THE INVENTION

(1) Field of the Invention

The present invention relates to a capacity control valve for a variable displacement compressor, and more particularly to a capacity control valve that regulates a flow of refrigerant discharged by a variable displacement compressor.

(2) Description of the Related Art

Automotive air conditioning systems employ a compressor to compress refrigerant gases in their refrigeration cycle. Since the compressor is driven by the automobile engine, the air conditioning system is unable to vary compressor rotation speed to control its output. To obtain a required cooling capacity without being restricted by the engine speed, the system uses a variable displacement compressor designed to be able to change the capacity (i.e., the amount of refrigerant that it can discharge) on its own.

In a variable displacement compressor, a wobble plate (swash plate) is fitted obliquely on the compressor's drive shaft, which is rotated by the engine. Rotation of this inclined wobble plate produces displacement of pistons that are linked to that plate, where the resulting piston strokes depends on the inclination angle of the wobble plate. This means that the compressor capacity (i.e., the amount of refrigerant being discharged from the compressor) can be varied by changing the wobble plate angle.

To control the wobble plate angle, part of the pressurized refrigerant is introduced into the gastight crank chamber of the compressor. Changing the crank chamber pressure creates a new state of balance between opposing pressures exerted on the both ends of each piston linked to the wobble plate, making it possible to vary the wobble plate angle steplessly.

To change the crank chamber pressure, a capacity control valve is installed either between the refrigerant outlet and the crank chamber or between the refrigerant inlet and the crank chamber. Capacity control valves are designed to open or close themselves in such a way that a certain level of differential pressure between their inlet and outlet will be maintained. More specifically, one can set a desired differential pressure by supplying a capacity control valve with an appropriate control current from an external power source. When the engine speed rises,

the capacity control valve raises the pressure of refrigerant supplied to the crank chamber so as to reduce the compressor capacity. When in turn the engine slows down, the capacity control valve decreases the crank chamber pressure so as to increase the compressor capacity. In this way, the amount of refrigerant discharged from the variable displacement compressor is regulated.

One method to control capacity of the above variable displacement compressors is disclosed in Unexamined Japanese Patent Application Publication No. 2001-107854 (Paragraphs (0035) to (0036), Figure 3). This literature describes a capacity control valve that regulates the flow of refrigerant being discharged from a variable displacement compressor.

According to Unexamined Japanese Patent Application Publication No. 2001-107854, the flow of refrigerant that is taken into the suction chamber is determined indirectly by detecting differential pressure between two pressure monitoring points with sensors. The capacity control valve controls the flow of refrigerant supplied from the discharge chamber to the crank chamber such that the intake flow rate will be maintained at a constant level, thereby regulating the flow of refrigerant discharged from the compressor.

The capacity control valve that controls a flow in the way described in Unexamined Japanese Patent Application Publication No. 2001-107854 needs sensors to

detect differential pressure, as well as a controller to control the capacity control valve accordingly. Those extra components push up the cost of variable displacement compressor.

Another noteworthy aspect of automobile air conditioning systems is what kind of refrigerant to choose for their refrigeration cycles. While HFC-134a, a chlorofluorocarbon alternative, is widely used for that purpose as of this point in time, the recent development of supercritical refrigeration cycles using, for example, carbon dioxide poses another challenge to the compressor design. The new refrigeration cycle requires refrigerant to function in a region that exceeds its critical temperature, hence the supercritical refrigeration. Let us think of a refrigeration cycle using carbon dioxide as refrigerant in which a capacity control valve is employed to control crank chamber pressure according to the compressor's discharge pressure. In this case, the differential pressure between the refrigerant outlet and crank chamber could become extremely high because the refrigerant has to be pressurized up to its supercritical region. This means that a high-power solenoid actuator will be needed to produce a sufficiently large force to deal with the high differential pressure, which leads to increased size and cost of the capacity control valve.

SUMMARY OF THE INVENTION

In view of the foregoing, it is an object of the present invention to provide a compact capacity control valve for use with a flow-controlled variable displacement compressor, which can be applied not only to ordinary refrigeration cycles using HFC-134a, but also to those using supercritical high-pressure refrigerant, without the needs for high-power solenoids or extra pressure sensors.

To solve the above-described problems, the present invention provides a capacity control valve that regulates a flow of refrigerant discharged from a variable displacement compressor. This capacity control valve comprises the following components formed in an integrated way: a first control valve that sets a specific cross-sectional area of a refrigerant passageway that leads to a suction chamber or a discharge chamber of the variable displacement compressor; a second control valve that senses differential pressure developed across the first control valve and controls a flow of refrigerant supplied to or coming out of the crank chamber of the variable displacement compressor in such a way that the differential pressure will be maintained at a specified level; and a solenoid unit that actuates the first control valve to set the cross-sectional area of the refrigerant passageway according to variations in a given external condition.

The above and other objects, features and

advantages of the present invention will become apparent from the following description when taken in conjunction with the accompanying drawings which illustrate preferred embodiments of the present invention by way of example.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of a variable displacement compressor.

FIG. 2 is a detailed sectional view of a capacity control valve for a variable displacement compressor according to the first embodiment.

FIG. 3 is a sectional view of a capacity control valve for a variable displacement compressor according to the second embodiment.

FIG. 4 is a sectional view of a capacity control valve for a variable displacement compressor according to the third embodiment.

FIG. 5 is a sectional view of a capacity control valve for a variable displacement compressor according to the fourth embodiment.

FIG. 6 is a sectional view of a capacity control valve for a variable displacement compressor according to the fifth embodiment.

FIG. 7 is a sectional view of a capacity control valve for a variable displacement compressor according to the sixth embodiment.

FIG. 8 is a sectional view of a capacity control

valve for a variable displacement compressor according to the seventh embodiment.

FIG. 9 is a sectional view of a capacity control valve for a variable displacement compressor according to the eighth embodiment.

FIG. 10 is a sectional view of a capacity control valve for a variable displacement compressor according to the ninth embodiment.

FIG. 11 is a sectional view of a capacity control valve for a variable displacement compressor according to the tenth embodiment.

FIG. 12 is a sectional view of a capacity control valve for a variable displacement compressor according to the eleventh embodiment.

FIG. 13 is a sectional view of a capacity control valve for a variable displacement compressor according to the twelfth embodiment.

FIG. 14 is a sectional view of a capacity control valve for a variable displacement compressor according to the thirteenth embodiment.

FIG. 15 is a sectional view of a capacity control valve for a variable displacement compressor according to the fourteenth embodiment.

FIG. 16 is a sectional view of a capacity control valve for a variable displacement compressor according to the fifteenth embodiment.

FIG. 17 is a sectional view of a capacity control

valve for a variable displacement compressor according to the sixteenth embodiment.

FIG. 18 is a sectional view of a capacity control valve for a variable displacement compressor according to the seventeenth embodiment.

FIG. 19 is a sectional view of a capacity control valve for a variable displacement compressor according to the eighteenth embodiment.

FIG. 20 is a sectional view of a capacity control valve for a variable displacement compressor according to the nineteenth embodiment.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Embodiments of the present invention will now be described below with reference to the accompanying drawings. The description will illustrate capacity control valves for use with a flow-controlled variable displacement compressor that is supposed to discharge refrigerant at a regulated flow rate.

FIG. 1 is a sectional view of a variable displacement compressor.

The explanation will begin with the overall structure of the variable displacement compressor 1 of FIG. 1.

The illustrated variable displacement compressor 1 is composed of the following three sections: a driving section 100 that receives drive power from a vehicle

engine (not shown); a refrigerant compressing section 200 including a gastight crank chamber; and a capacity controlling section 300 that controls discharge capacity. The variable displacement compressor 1 has an outlet port 1a, which is connected to a condenser (or gas cooler) 3 through a high-pressure refrigerant line 2. The refrigerant is then routed from the condenser 3 to an expansion valve 4, an evaporator 5, and a low-pressure refrigerant line 6 in that order, and finally returns to the inlet port 1b of the variable displacement compressor 1, thus forming a closed circuit for refrigeration cycle.

The driving section 100 is constructed such that the rotational power of the engine can be transmitted from a drive pulley 13 to a bracket 14, and then to a rotating shaft 12 that protrudes out of a front housing 11. In the refrigerant compressing section 200, the crank chamber 15 is formed as a closed space surrounded by a front housing 11 and a cylinder block 16. The rotating shaft 12 is rotatably installed in the crank chamber 15, across the length of the front housing 11 and cylinder block 16.

The drive pulley 13 is rotatably supported by an angular bearing 17 at the front housing 11. A drive belt (not shown) is installed around the circumference of the drive pulley 13. The bracket 14, which rotates together with the drive pulley 13, is coupled to one end of the rotating shaft 12 that protrudes from the front housing 11. As can be seen, the rotation of the vehicle engine is

directly transmitted to the variable displacement compressor 1, with no intervening clutch mechanisms (e.g., electromagnetic clutch) between them.

To seal off the crank chamber 15 from the exterior space of the refrigerant compressing section 200, a lip seal 18 is placed between the front housing 11 and the front portion of the rotating shaft 12. The rotational support member 19 is fixed to the rotating shaft 12 in the crank chamber 15. A swash plate 20 is supported in such a way that it can be inclined at an oblique angle to the axis of the rotating shaft 12. The swash plate 20 has a guide pin 22, whose spherical top portion is engaged with a support arm 21 that is mounted on the rotational support member 19 in a protruding manner. This linkage between the support arm 21 and guide pin 22 enables the swash plate 20 to rotate together with the rotating shaft 12.

Interposed between the rotational support member 19 and swash plate 20 is an inclination-reducing spring 23, which urges the swash plate 20 in the direction that its inclination angle is reduced. The maximum swash plate angle is restricted by an inclination-limiting protrusion 20a of the swash plate 20 itself, which juts out toward the rotational support member 19.

The rotating shaft 12 is rotatably supported at its rear end by a radial bearing 24 that is mounted at a central axis position of the cylinder block 16.

The cylinder block 16 has a plurality of cylinder

bores 16a formed in a manner that they pass through the cylinder block 16. Those cylinder bores 16a house a plurality of single-headed pistons 25 (hereafter, "pistons"), one for each. The swash plate 20 engages with the head of each piston 25 via shoes 26, which permits rotational motion of the swash plate 20 to be converted into reciprocating motion of the pistons 25. Placed between the rotational support member 19 and front housing 11 is a thrust bearing 28, which receives reaction forces that are caused by the compression of refrigerant and act on the rotational support member 19 via the pistons 25 and swash plate 20.

The capacity controlling section 300 is attached to the refrigerant compressing section 200, with a valve plate 27 separating them. The capacity controlling section 300 is made up of a rear housing 31 located next the valve plate 27 and a capacity control valve 30 (described later) installed and secured in a predetermined position in the rear housing 31. The rear housing 31 provides the following separate cavities formed immediately beside the valve plate 27: suction chambers 32, discharge chambers 33, and a communication passage 34. The suction chambers 32 are cavities at suction pressure Ps. The discharge chambers 33 at discharge pressure PdH receive the refrigerant compressed by the refrigerant compressing section 200. The communication passage 34 communicates with the crank chamber, and hence is at crank chamber

pressure Pc. In addition, the rear housing 31 provides an outlet port 1a and an inlet port 1b of the variable displacement compressor 1, as well as a housing cavity 35 for accommodating the capacity control valve 30. Further, the rear housing 31 has several communication holes 36 to 39 formed in its body. The first communication hole 36 connects the inlet port 1b with the suction chambers 32. The second communication hole 37 connects the housing cavity 35 with the communication passage 34, which further leads to the crank chamber 15. The housing cavity 35 can also communicate with the discharge chambers 33 through the third communication hole 38. The fourth communication hole 39 permits the housing cavity 35 to communicate with the outlet port 1a of the variable displacement compressor 1.

A suction relief valve 32v is placed at each cylinder port connecting to the suction chamber 32, on the side of the valve plate 27 adjacent to the cylinder bores 16a. A discharge relief valve 33v is placed similarly at each cylinder port connecting to its corresponding discharge chambers 33, but on the opposite side of the valve plate 27, remote from the cylinder bores 16a. The suction chambers 32, one for each cylinder bore 16a, communicate with each other in the rear housing 31, as well as with the first communication hole 36. Likewise, the discharge chambers 33 communicate with each other in the rear housing 31, as well as with the third

communication hole 38. As the pistons 25 reciprocate, the refrigerant gas in the suction chamber 32 is sucked into each cylinder bore 16a through its corresponding suction relief valve 32v and then discharged from those cylinder bores 16a to their corresponding discharge chambers 33 through respective discharge relief valves 33v.

While not shown in FIG. 1, there is a fixed orifice between the crank chamber 15 and suction chambers 32 to release the refrigerant from the crank chamber 15 to the suction chambers 32.

The following will now describe several specific examples of the proposed capacity control valve for a variable displacement compressor.

First Embodiment

FIG. 2 is a detailed sectional view of a capacity control valve for a variable displacement compressor according to the first embodiment.

This capacity control valve 30 is made up of a first control valve 30A, a second control valve 30B, and a solenoid unit 30C.

The first control valve 30A has two ports 41 and 42 formed on its body 40. One port 41 receives a discharge pressure PdH from the discharge chambers 33 through the third communication hole 38 of the rear housing 31 shown in FIG. 1. The other port 42 outputs refrigerant at discharge pressure PdL that has been reduced at the first

control valve 30A, for delivery through the fourth communication hole 39 and then the high-pressure refrigerant line 2. Bored between those two ports 41 and 42 is a valve hole 45 for communication of refrigerant, the upstream edge of which is intended to function as a first valve seat 45a. In an upstream space adjacent to the first valve seat 45a, a ball-shaped valve element (ball valve element) 46 is placed opposite to the first valve seat 45a. This ball valve element 46 is referred to herein as the first valve element. The valve hole space communicating with the port 41 accommodates a coil spring 48 that urges the ball valve element 46 in the direction that it closes the passage, and the amount of that spring load can be adjusted by turning an adjustment screw 47, which is screwed into the body 40.

The downstream side of the ball valve element 46 is in contact with one end of a shaft 49 that extends in the axial direction of the solenoid unit 30C through the valve hole of the first valve seat 45a. This shaft 49 is supported by a bearing 50a formed in the body 40, and the bearing 50a has a communication hole 50b to equalize the inside pressure of the solenoid unit 30C with the discharge pressure PdL.

The solenoid unit 30C contains a solenoid coil 51 that has a cylindrical cavity, in which a sleeve 52 is fitted. As the fixed core of the solenoid actuator, a core 53 is pressed into the sleeve 52 through its opening end

adjacent to the first control valve 30A. The sleeve 52 also contains a plunger 54 that can slide in its axial direction while being urged by a coil spring 55 in the downward direction as viewed in FIG. 2. The plunger 54 is fixed to the lower end (as viewed in FIG. 2) of the shaft 49 running coaxially through the core 53. This arrangement permits the capacity control valve 30 to operate as follows. When the solenoid coil 51 is in de-energized state, the plunger 54 is set away from the core 53 due to the force of the coil spring 55, causing the shaft 49 extending from the plunger 54 to lose contact with the ball valve element 46. As a result, the first control valve 30A becomes fully closed because the freed ball valve element 46 is seated on the first valve seat 45a, being urged by another coil spring 48. When, on the other hand, the solenoid coil 51 is energized, the plunger 54 attracted by the magnetized core 53 will pushes the ball valve element 46 via the shaft 49 in the valve-opening direction (i.e., in the direction that the valve element will leave its corresponding valve seat). The ball valve element 46 thus moves, and the amount of this movement, or the valve lift (or openness), is proportional to the electrical current being supplied to the solenoid coil 51. This means that the control current given to the solenoid coil 51 determines the cross-sectional area of the refrigerant passageway that the first control valve 30A provides. In other words, the first control valve 30A

functions as a variable orifice, which changes its cross-sectional size as specified by the control current to allow the discharged refrigerant to pass through it.

The solenoid unit 30C described above is intended, not for directly controlling high-pressure refrigerant flow, but for controlling the first control valve 30A so that a small differential pressure will be produced depending on the discharge flow rate Qd of the refrigerant passing therethrough. Since only a small power is needed to achieve the purpose, it is possible to reduce the size of the solenoid unit 30C.

The second control valve 30B has a body 40a, which is screwed to the body 40 of the first control valve 30A so that the two valves 30A and 30B are stacked in series. The body 40a has two ports 43 and 44. One port 43 is used to apply controlled pressure Pc to the crank chamber, and the other port 44 is used to introduce discharge pressure PdL that has been reduced at the first control valve 30A. The body 40a also has an opening at its bottom end, which communicates with the port 41 to receive discharge pressure PdH of the discharge chambers 33 through a communication hole 47a formed on an adjustment screw 47. Between this opening and the port 43, a second valve seat 56 is formed as an integral part of the body 40a. Placed opposite to this second valve seat 56 in the port 43 is a second valve element 57. The second valve element 57 is a taper-shaped member that is integrally formed with a

cylindrical piston 58, where the piston 58 can move in its axial direction within a cylinder that is bored on the axis of the body 40a. A coil spring 60 is installed at the upper end portion of the piston 58 as viewed in FIG. 2, which urges the second valve element 57 in the valve-closing direction. This spring load depends on how much an adjustment screw 59 is screwed into the body 40a. The adjustment screw 59 has a through hole 59a at its central position, and this through hole 59a serves as a passage for introducing the reduced discharge pressure PdL from the port 44 to the space above the piston 58. The second valve element 57 and piston 58 thus receive different pressures at their both endfaces apart in the direction of their axis. That is, the second valve element 57 receives discharge pressure PdH from its nearest port 41, while the piston 58 receives discharge pressure PdL from its nearest port 44. Their differential pressure ΔP determines the lift of the second valve element 57. More specifically, differential pressure ΔP is produced when refrigerant flows through a passage with a certain cross-sectional area that is determined by the first control valve 30A. Then the second control valve 30B functions as a constant differential pressure valve that controls the amount of refrigerant flowing into the crank chamber 15 in such a way that the above differential pressure ΔP will be maintained at a constant level.

Several O-rings are provided around the periphery

of the capacity control valve 30. They include: an O-ring 29a to seal up the gap between the ports 44 and 43, another O-ring 29b between the ports 43 and 41, yet another O-ring 29c between the ports 41 and 42, still another O-ring 29d between the port 42 and solenoid unit 30C, and yet another O-ring 29e to seal the solenoid unit 30C off from the surrounding atmosphere.

The variable displacement compressor 1 described above operates as follows. When the rotating shaft 12 is driven by the engine power, the swash plate 20 begins to wobble while turning around that rotating shaft 12. This wobbling produces reciprocating motion of the pistons 25 that are linked to the outer regions of the swash plate 20, which causes refrigerant to be sucked from the suction chambers 32 into the cylinder block 16. The refrigerant is thus compressed and discharged toward the discharge chambers 33.

Suppose here that the solenoid unit 30C is in de-energized state. Since the first control valve 30A is fully closed in this state, the refrigerant discharged to the discharge chambers 33 is entered to the crank chamber 15 in its entirety via the second control valve 30B. This causes the variable displacement compressor 1 to run in the minimum capacity mode.

When a predetermined amount of control current is supplied to the solenoid unit 30C, the first control valve 30A gives a predetermined openness (valve lift) associated

with that control current. The first control valve 30A now acts as an orifice with a certain cross-sectional size, allowing a flow of refrigerant through the high-pressure refrigerant line 2 leading to the condenser 3. This develops a certain amount of differential pressure ΔP (=PdH-PdL) across the orifice, depending on the actual discharge flow rate Qd of the refrigerant passing through it.

In the second control valve 30B, its second valve element 57 and piston 58 are responsive to the differential pressure ΔP across the first control valve 30A, which is functioning here as an orifice. The second control valve 30B controls the flow of refrigerant from the discharge chambers 33 to the crank chamber 15 in such a way that the differential pressure ΔP will be maintained at a constant level. This control action may vary the capacity of the variable displacement compressor 1 as needed, so as to regulate the flow of refrigerant being discharged therefrom.

The flow rate of refrigerant discharged from the variable displacement compressor 1 is determined depending on how much refrigeration capacity is required in the present refrigeration cycle. Actually, the refrigeration capacity is calculated from various parameters, which include: engine rotation speed, vehicle speed, accelerator pedal position, indoor and outdoor temperatures, set temperatures, and monitoring signals supplied from various

temperature and pressure sensors. The amount of the electrical current that energizes the solenoid coil 51 is determined on the basis of this calculation result.

Suppose here that the engine rotation rises and the discharge flow rate of refrigerant is increased accordingly. This develops an increased differential pressure ΔP across the first control valve 30A. In response to ΔP, the second control valve 30B lifts its valve element, so that more refrigerant will be supplied from the discharge chambers 33 to the crank chamber 15. As a result, pressure Pc in the crank chamber 15 rises, and the variable displacement compressor 1 is thus controlled in an output-reducing condition. The variable displacement compressor 1 now operates with a smaller discharge capacity, suppressing the discharge flow rate of refrigerant, and thus reducing the differential pressure ΔP. In this way, the discharge flow rate Qd of refrigerant is regulated by controlling the second control valve 30B so that the differential pressure across the orifice (i.e., the first control valve 30A being configured as a proportional solenoid valve) will be maintained at a constant level.

The engine rotation may in turn drops. This decreases the flow rate of discharged refrigerant and reduces the differential pressure across the first control valve 30A accordingly. The refrigerant discharge pressure PdH falls, and thus the second control valve 30B operates

in such a way as to reduce the refrigerant flow from the discharge chambers 33 to the crank chamber 15. Pressure Pc in the crank chamber 15 falls accordingly, which causes the variable displacement compressor 1 to operate in a capacity-increasing condition, thus recovering the discharge. In this way, the discharge flow rate Qd of refrigerant is maintained at the constant level.

As can be seen from the above description, the present invention provides a capacity control valve 30 for use with a variable displacement compressor. This capacity control valve 30 is composed of a first control valve 30A that functions as a variable orifice controlled by a solenoid unit 30C and a second control valve 30B that controls the pressure in the crank chamber 15 so as to maintain a constant differential pressure across the variable orifice. The present invention combines those components in an integrated way, thus providing a compact, space-saving design for the capacity control functions (i.e., regulating the flow rate Qd of refrigerant discharged from the variable displacement compressor 1).

Second Embodiment

FIG. 3 is a sectional view of a capacity control valve for a variable displacement compressor according to the second embodiment of the invention. Since many of the valve components shown in FIG. 3 are identical or similar to those discussed in FIG. 2, the same reference numerals

are used in FIG. 3 to designate such components, and the following section will not provide details about them.

The illustrated capacity control valve 30 of the second embodiment resembles that of the first embodiment (FIG. 2) in that they share the same basic structure of their first control valve 30A and second control valve 30B, as well as in that the two valves 30A and 30B are stacked in series. The second embodiment is, however, different from the first embodiment in that the ball valve element 46 of its first control valve 30A is arranged in such a way that it will allow more refrigerant to pass through when it is displaced following the stream of refrigerant. In other words, the ball valve element 46, or the first valve element, is placed on the downstream side with respect to the first valve seat 45a. To make this arrangement possible, the plunger 54 and core 53 have to swap their positions in the solenoid unit 30C.

The first control valve 30A stays in a fully closed position when the solenoid unit 30C is not energized, because the ball valve element 46 is seated on the first valve seat 45a due to the force of a coil spring 55 installed between the plunger 54 and core 53. Accordingly, the refrigerant coming into the port 41 at discharge pressure PdH is led to the crank chamber 15 in its entirety through the second control valve 30B, meaning that the variable displacement compressor 1 now operates in the minimum capacity condition.

When a predetermined amount of control current is supplied to a solenoid coil 51 of the solenoid unit 30C, the plunger 54 is attracted by the core 53 and stops at the point where the attraction force associated with that control current comes into balance with the urging force of the coil spring 55. In this state, the ball valve element 46 is lifted, keeping in contact with the shaft 49 due to the force of the coil spring 48, and the consequent gap serves as an orifice with a designated size.

Variations in the engine speed affect the discharge flow from the variable displacement compressor. In this situation, the capacity control valve 30 of the second embodiment operates in the same way as in the first embodiment described earlier in FIG. 2.

Third Embodiment

FIG. 4 is a sectional view of a capacity control valve for a variable displacement compressor according to a third embodiment of the invention. Since many of the valve components shown in FIG. 4 are identical or similar to those discussed in FIGS. 2 and 3, the same reference numerals are used in FIG. 4 to designate such components, and the following section will not provide details about them.

Recall that a ball valve element 46 is used as the first valve element in the first and second embodiments (FIGS. 2 and 3). However, the illustrated capacity control

valve 30 of the third embodiment is different in that a taper-shaped valve element 61 is placed on the upstream side with respect to the first valve seat 45a while receiving a force in the valve-opening direction. Another difference is that it eliminates the port 44, which is employed in the first and second embodiments to introduce discharge pressure PdL into the second control valve 30B. Instead, the illustrated capacity control valve 30 has a communication hole 62 formed in the body 40 to serve the purpose. To make this arrangement possible, the port 41 for discharge pressure PdH and port 42 for discharge pressure PdL have swapped their positions in the third embodiment. Yet another difference is that the crank chamber 15 receives discharge pressure PdL after orifice.

More specifically, the first control valve 30A has a port 41 formed in its body 40 to receive discharge pressure PdH from the discharge chambers 33. It has another port 42 formed in the same body 40 to supply the high-pressure refrigerant line 2 with discharge pressure PdL that is reduced by the first control valve 30A. A valve hole 45 is bored for communication between those two ports 41 and 42, and its upstream-side edge is intended to function as a first valve seat 45a. A taper-shaped valve element 61 is placed in an upstream-side space, opposite to the first valve seat 45a. This valve element 61 is referred to herein as a first valve element. A flange 61a is formed as an integral part of the first valve element

61, on its circumference remote from the first valve seat 45a.

The flange 61a retains one end of a coil spring 48 that is placed around the first valve element 61 against the first valve seat 45a. This coil spring 48 urges the first valve element 61 in the direction that the valve will open. The valve element 61 is also coupled to an end of a shaft 49 that extends from the solenoid unit 30C in its axial direction. When the solenoid unit 30C is in de-energized state, the coil spring 55 makes the first valve element 61 sit on the first valve seat 45a. The shaft 49 is supported by a bearing 50a at its middle portion adjacent to the first control valve 30A, as well as by another bearing 50c at its bottom end. The bottom-end bearing 50c has been pressed into the central bore of the core 53.

The second control valve 30B is coupled in series with the first control valve 30A, the space above the piston 58 being closed by a lid 59b. Its body 40 has a communication hole 62 to communicate that space with the port 41, through which discharge pressure PdH acts on the back face of the piston 58. This arrangement of the third embodiment reduces the number of ports that should be created on the body 40, thus making it easier to manufacture the capacity controlling section 300 of a variable displacement compressor 1. It also eliminates some O-rings that are required when fitting the capacity

control valve 30 into the housing cavity 35 of the variable displacement compressor 1.

As can be seen from the above description, the illustrated capacity control valve 30 contains a first control valve 30A composed of a first valve element 61 and first valve seat 45a, the first valve element 61 being a taper-shaped valve element located in an upstream-side space adjacent to the first valve seat 45a. Here, the first control valve 30A sets a certain cross-section area for the refrigerant passageway in accordance with how much the solenoid unit 30C is energized. The second control valve 30B is responsive to differential pressure developed across the first control valve 30A to control the flow rate of refrigerant supplied from the discharge chambers 33 to the crank chamber 15. In the way described above, the capacity control valve 30 regulates the flow rate Qd of refrigerant that the variable displacement compressor 1 discharges.

Fourth Embodiment

FIG. 5 is a sectional view of a capacity control valve for a variable displacement compressor according to a fourth embodiment of the invention. Since many of the valve components shown in FIG. 5 are identical or similar to those discussed in FIG. 2, the same reference numerals are used in FIG. 5 to designate such components, and the following section will not provide details about them.

Compared with the first embodiment discussed earlier in FIG. 2, the capacity control valve 30 of the fourth embodiment is distinct in the following points. First, its first control valve 30A employs a spool-shaped valve element 63 as a first valve element. Second, its second control valve 30B uses a taper-shaped valve element 64 as a second valve element. Third, as the counterpart of the spool-shaped valve element 63 (or first valve element), a first valve seat 63a is provided as an integral part of the second valve element in the second control valve 30B. This first valve seat 63a is designed to set a required cross-section area for refrigerant passage while moving together with the second valve element.

More specifically, the second control valve 30B has a second valve seat 56 and its corresponding second valve element 64 with a tapered shape. The second valve seat 56 is formed as an integral part of the body 40, in the middle of a refrigerant passageway between two ports 41 and 43, the former receiving refrigerant from the discharge chambers 33 and the latter delivering refrigerant to the crank chamber 15. Opposite the second valve seat 56, the second valve element 64 is located in an upstream-side space, where discharge pressure PdH is available. The second valve element 64 is urged by a coil spring 66 in the valve-opening direction. Integrally formed with this second valve element 64 is a pressure responsive member 64a, whose base portion detects

differential pressure between two different discharge pressures PdH and PdL. The pressure responsive member 64a is installed inside the body 40 in a manner that it can come in contact with or move away from the second valve seat 56 according to the differential pressure acting thereon. The pressure responsive member 64a has a central cavity around its axis, the bottom end of which is open. The pressure responsive member 64a has also a hole 64b in its upper portion, which allows the discharge pressure PdH in the port 41 to reach the central cavity.

The first control valve 30A, on the other hand, has a first valve seat 63a formed around the rim of the bottom opening of the pressure responsive member 64a, which operates together with a spool-shaped valve element (or a first valve element) 63 located below the bottom opening. The first valve seat 63a and first valve element 63 set an appropriate cross-sectional area for a passageway that delivers refrigerant from one port 41 to another port 42 via the hole 64b of the pressure responsive member 64a.

The spool-shaped valve element 63, or the first valve element, is integrally formed with a pressure responsive piston 63p having the same cross-sectional area as the valve hole of the first valve seat 63a. A flange 63b is formed around this valve element 63, on a downstream-side portion remote from the first valve seat 63a. This flange 63b is used to receive a force of a coil

spring 48, which urges the valve element 63 in the valve-opening direction. Another coil spring 60 is disposed between the pressure responsive piston 63p and the pressure responsive member 64a of the second valve element 64. The pressure responsive piston 63p is slidably supported by a plug 40b, which seals the bottom of the body 40. The pressure responsive piston 63p may also be pressed upward by a shaft 49. This shaft 49 extends from the solenoid unit 30C in its axial direction and reaches the bottom endface of the pressure responsive piston 63p. A pressure balancing hole 65 is bored through the pressure responsive piston 63p to introduce a back pressure from the upstream-side cavity adjacent to the first valve seat 63a. This structure permits the discharge pressure PdH from the port 41 to act equally on both the bottom end of the pressure responsive piston 63p and the top end of the spool-shaped valve element 63. Since those two opposing forces cancel each other out, the discharge pressure PdH never disturbs the solenoid unit 30C when it controls the position of the valve element 63.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 upward as viewed in FIG. 5, making the spool-shaped valve element 63 fit into the central opening of the pressure responsive member 64a. The first control valve 30A is fully closed in this state, while the

second control valve 30B fully opens itself in attempt to obtain a predetermined differential pressure between discharge pressures PdH and PdL acting on the pressure responsive member 64a.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 5. This movement of the shaft 49 allows the spool-shaped valve element 63 to come out of the first valve seat 63a and maintain a certain amount of gap between itself and the first valve seat 63a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a of the second valve element 64 receives differential pressure between discharge pressures PdH and PdL, which moves the second valve element 64 so that the differential pressure will become a predetermined level. With this movement of the second valve element 64, the second control valve 30B controls the refrigerant being delivered from its port 43 to the crank chamber 15.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. The increased differential pressure causes the second valve element 64 to move in the valve-opening direction, so that the second control valve 30B supplies more refrigerant into the crank chamber 15. As a result of this control

action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flowing into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Fifth Embodiment

FIG. 6 is a sectional view of a capacity control valve for a variable displacement compressor according to a fifth embodiment of the invention. Since many of the valve components shown in FIG. 6 are identical or similar to those discussed in FIG. 5, the same reference numerals are used in FIG. 6 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of the fifth embodiment is similar to that of the fourth embodiment (FIG. 5) in that their first control valve 30A employs a spool-shaped valve element 63 as a first valve element. The fifth embodiment, however, is different from the fourth embodiment in that the port 41 for discharge pressure PdH and port 42 for discharge pressure PdL have swapped their positions. While the second control valve

30B of the fourth embodiment has a taper-shaped valve element 64 as a second valve element, the fifth embodiment employs a ball valve element 67 for that purpose. This ball valve element (or the second valve element) 67 is located downstream with respect to the second valve seat 56, while being urged in the valve-opening direction by a stem that extends through the valve hole and the first control valve 30A.

More specifically, the second control valve 30B has a second valve seat 56 formed as an integral part of its body 40, and a ball valve element 67 is located in a downstream-side space adjacent to the second valve seat 56. The ball valve element 67 is urged by a coil spring 60 in the valve-closing direction, the spring load of which can be adjusted by turning an adjustment screw 59. The adjustment screw 59 has a through hole 59a in its central portion, and this through hole 59a serves as a port 43 for delivery of refrigerant to the crank chamber 15.

The first control valve 30A has a first valve seat 63a at the bottom end of a pressure responsive member 64a. The pressure responsive member 64a is integrally formed with a shaft 68 that extends in the axial direction of the second control valve 30B, passing through the valve hole of same. The upper end of this shaft 68 is in contact with the ball valve element 67 of the second control valve 30B.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit

30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction as viewed in FIG. 6, making the spool-shaped valve element 63 fit into the central opening of the pressure responsive member 64a. The first control valve 30A is fully closed in this state, while the second control valve 30B is fully opened because of the differential pressure that acts on the pressure responsive member 64a.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 6. This movement of the shaft 49 allows the spool-shaped valve element 63 to come out of the first valve seat 63a and maintain a certain amount of gap between itself and the first valve seat 63a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a receives differential pressure between discharge pressures PdH and PdL, which moves the ball valve element 67 so that the differential pressure will become a predetermined level. With this movement of the ball valve element 67, the second control valve 30B controls the flow rate of the refrigerant being delivered from its port 43 to the crank chamber 15.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the

increased differential pressure, the ball valve element 67 gives a greater openness, so that the second control valve 30B supplies more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flowing into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Sixth Embodiment

FIG. 7 is a sectional view of a capacity control valve for a variable displacement compressor according to a sixth embodiment of the invention. Since many of the valve components shown in FIG. 7 are identical or similar to those discussed in FIG. 2, 4, or 6, the same reference numerals are used in FIG. 7 to designate such components, and the following section will not provide details about them.

As in the third embodiment (FIG. 4), the capacity control valve 30 of the sixth embodiment is different from that of the fifth embodiment (FIG. 6) in that its first

control valve 30A employs a taper-shaped valve element 61 as a first valve element, and in that the valve element 61 is located in an upstream-side space adjacent to the first valve seat 63a, being urged in the valve-opening direction.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 upward as viewed in FIG. 7, making the taper-shaped valve element 61 sit on the first valve seat 63a. The first control valve 30A is fully closed in this state, while the second control valve 30B is fully opened because of the differential pressure that acts on the pressure responsive member 64a.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 7. This movement of the shaft 49 allows the taper-shaped valve element 61 to leave the first valve seat 63a and maintain a certain amount of gap between itself and the first valve seat 63a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a receives differential pressure between discharge pressures PdH and PdL, which moves the ball valve element 67 so that the differential pressure will become a predetermined level. With this movement of the ball valve element 67, the second control valve 30B controls the flow rate of the

refrigerant being delivered from its port 43 to the crank chamber 15.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure, the ball valve element 67 gives a greater openness, allowing the second control valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flowing into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Seventh Embodiment

FIG. 8 is a sectional view of a capacity control valve for a variable displacement compressor according to a seventh embodiment of the invention. Since many of the valve components shown in FIG. 8 are identical or similar to those discussed in FIG. 6 or 7, the same reference numerals are used in FIG. 8 to designate such components,

and the following section will not provide details about them.

The illustrated capacity control valve 30 of the seventh embodiment is different from that of the sixth embodiment (FIG. 7) only in that its first valve element 61 is designed to cancel the back pressure in order to prevent the discharge pressure PdH from affecting operation of the first control valve 30A. This concept is what has been described in the fifth embodiment (FIG. 6). The capacity control valve 30 of the seventh embodiment operates basically in the same as that of the sixth embodiment.

More specifically, when the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction as viewed in FIG. 8. This makes the taper-shaped valve element 61 sit on the first valve seat 63a. Accordingly, the first control valve 30A is fully closed, while the second control valve 30B is fully opened.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 8. This movement of the shaft 49 allows the taper-shaped valve element 61 to leave the first valve seat 63a and maintain a certain amount of gap between itself and the first valve seat 63a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second

control valve 30B, the pressure responsive member 64a receives differential pressure between discharge pressures PdH and PdL, which moves the ball valve element 67 so that the differential pressure will become a predetermined level. With this movement of the ball valve element 67, the second control valve 30B controls the flow rate of the refrigerant being delivered from its port 43 to the crank chamber 15.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure, the ball valve element 67 gives a greater openness, allowing the second control valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flowing into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Eighth Embodiment

FIG. 9 is a sectional view of a capacity control valve for a variable displacement compressor according to an eighth embodiment of the invention. Since many of the valve components shown in FIG. 9 are identical or similar to those discussed in FIG. 2 or 5, the same reference numerals are used in FIG. 9 to designate such components, and the following section will not provide details about them.

The illustrated capacity control valve 30 of the eighth embodiment resembles that of the fourth embodiment (FIG. 5) in that both of them use a taper-shaped valve element 64 as their second valve element. The eighth embodiment is, however, different from the fourth embodiment in that its first valve seat 45a is not designed to move, but is constructed as an integral part of the body 40 of the first control valve 30A. Another difference is that a plurality of ball valve elements 46 are employed to serve the function of the first valve element.

More specifically, the first control valve 30A is constructed as follows. The body 40 has a plurality of valve holes 45 bored along a circle that is concentric with a cross section of the body 40 itself, the bottom-end edge of each hole serving as a first valve seat 45a. A ball valve element 46 is placed at a downstream-side space adjacent to each first valve seat 45a. Those ball valve elements 46 sit on the downstream-side surface of a

support member 70, which is urged by a coil spring 60 in the downward direction as viewed in FIG. 9. The support member 70 also receives a force of a coil spring 55 in the solenoid unit 30C, via a plunger 54 and shaft 49, which acts in the upward direction as viewed in FIG. 9.

The second control valve 30B, on the other hand, has a pressure responsive member 64a, which is urged by a coil spring 66 upward as viewed in FIG. 9. Since this pressure responsive member 64a is integrally formed with a second valve element 64, the urging force of the coil spring 66 also acts on the second valve element 64 in the valve-closing direction. The pressure responsive member 64a, in combination with the second valve element 64, is supposed to be responsive to differential pressure ΔP between two different discharge pressures PdH and PdL, which are observed on the upstream end and downstream end of the first control valve 30A, respectively.

When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction as viewed in FIG. 9. This makes the ball valve elements 46 fit with corresponding first valve seats 45a, and thus the first control valve 30A is fully closed. With the refrigerant at discharge pressure PdH present in the port 41, the maximum differential pressure acts on the pressure responsive member 64a, making the second control valve 30B fully open. The variable displacement compressor 1 thus operates in the

minimum capacity condition.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 9. This movement of the shaft 49, in conjunction with the force of the coil spring 60, allows the support member 70 to follow in the same direction while keeping in contact with the shaft 49. Each ball valve element 46 thus leaves the corresponding first valve seat 45a and maintains a certain amount of gap between itself and that valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a adjacent to the second valve element 64 receives differential pressure between discharge pressures PdH and PdL, which moves the second valve element 64 so that the differential pressure will become a predetermined level. This movement of the second valve element 64 controls the flow rate of the refrigerant at discharge pressure PdH that flows through the second control valve 30B, from one port 41 to another port 43.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure acting on the pressure responsive member 64a, the second valve element 64 in the second control valve 30B is moved in the direction that it gives a greater openness, allowing the second control

valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B reduces the refrigerant into the crank chamber 15 because the pressure responsive member 64a impels the second valve element 64 in the valve-closing direction. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Ninth Embodiment

FIG. 10 is a sectional view of a capacity control valve for a variable displacement compressor according to a ninth embodiment of the invention. Since many of the valve components shown in FIG. 10 are identical or similar to those discussed in FIG. 9, the same reference numerals are used in FIG. 10 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of this embodiment differs from that of the eighth embodiment (FIG. 9) in its first control valve 30A, particularly in the structure of its first valve element and first valve seat.

More specifically, the first control valve 30A is

constructed as follows. The body 40 has a doughnut-shaped valve hole 45 hollowed along a circle that is concentric with a cross section of the body 40 itself, and the bottom-end edge of that hole is supposed to serve as a first valve seat 45a. It should be noted that the doughnut-shaped valve hole 45 does not go through the floor of the body 40 in its entire circumference, but some middle part of the floor is left unhollowed at a few places on its circumference. This is necessary because the central portion of the floor should still be connected to the body 40 in order to house a pressure responsive member 64a in that portion. As the counterpart of the first valve seat 45a, a flat valve element 71 is disposed on the downstream side, together with a plug 40b that supports the flat valve element 71 in a way that it can slide in the axial direction.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction as viewed in FIG. 10, making the flat valve element 71 abut on the first valve seat 45a. The first control valve 30A is fully closed in this state. With the refrigerant at discharge pressure PdH present in the port 41, the maximum differential pressure acts on the pressure responsive member 64a, making the second control valve 30B fully open. The variable displacement compressor 1 thus operates in

the minimum capacity condition.

When the solenoid unit 30C is energized, the shaft 49 moves downward as viewed in FIG. 10. This movement of the shaft 49, in conjunction with the force of a coil spring 60, allows the flat valve element 71 to follow in the same direction while keeping in contact with the shaft 49. The flat valve element 71 thus leaves the first valve seat 45a and maintains a certain amount of gap between itself and that valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a adjacent to the second valve element 64 receives differential pressure between discharge pressures PdH and PdL, which moves the second valve element 64 so that the differential pressure will become a predetermined level. This movement of the second valve element 64 controls the flow rate of the refrigerant at discharge pressure PdH that flows through the second control valve 30B, from one port 41 to another port 43.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure acting on the pressure responsive member 64a, the second valve element 64 in the second control valve 30B is moved in the direction that it gives a greater openness, allowing the second control

valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B reduces the refrigerant into the crank chamber 15 because its pressure responsive member 64a impels the second valve element 64 in the valve-closing direction. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Tenth Embodiment

FIG. 11 is a sectional view of a capacity control valve for a variable displacement compressor according to a tenth embodiment of the invention. Since many of the valve components shown in FIG. 11 are identical or similar to those discussed in FIG. 2 or 4, the same reference numerals are used in FIG. 11 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of this embodiment differs from that of the first embodiment (FIG. 2) in several points. The most prominent difference is that the tenth embodiment uses, in its first control valve 30A, a

diaphragm 72 to detect differential pressure between upstream and downstream.

More specifically, in a central region of the body 40, a cylinder 40c is formed as an integral part of the body 40, and the inner cavity of that cylinder 40c serves as a valve hole 45 to interconnect two ports 41 and 42. The bottom end of the cylinder 40c functions as a first valve seat 45a for the first control valve 30A. In the downstream-side space communicating with the port 42, a taper-shaped valve element 61 is placed opposite the first valve seat 45a. This taper-shaped valve element 61, integrally formed with the plunger 54 of the solenoid unit 30C, has a circumferential groove 61b around its round side surface at the boundary portion where the plunger 54 is joined. The groove 61b receives a piston ring 74, which permits the plunger 54 to be slidably supported on the inner wall of the sleeve 52, as well as centering the taper-shaped valve element 61 on the axis of the sleeve 52.

In the second control valve 30B, on the other hand, a valve hole is bored to allow a port 41 to communicate with another port 43, the bottom end of which is supposed to function as a second valve seat 56. In the upstream-side space adjacent to the second valve seat 56, a taper-shaped second valve element 64 is placed. Integrally formed on top of this second valve element 64 are a shaft 64c and a piston 64d. This piston 64d has the same outer diameter as the valve hole of the second valve seat 56.

The endface of the piston 64d remote from the second valve element 64 receives, through a communication hole 62, discharge pressure PdH in the port 41, so that the second valve element 64 can be driven with nothing but differential pressure between discharge pressures PdH and PdL, without being affected by the absolute value of discharge pressure PdH. The second valve element 64 is integrally formed also with a base member 64e, which is larger in diameter than the second valve element 64 and has a hole 64b to introduce discharge pressure PdH from the port 41 to the inner cavity of the cylinder 40c.

A sliding member 73 is provided around the outer surface of the cylinder 40c in the body 40 in a way that it can move in the vertical direction as viewed in FIG. 11. This sliding member 73 is connected with the inner surface of the bodies 40 and 40a via a diaphragm 72, which is a doughnut-shaped sheet with a center hole. The outer circumference of the diaphragm 72 is clamped between two bodies 40 and 40a, the latter 40a being pressed into the former 40. The inner circumference of the diaphragm 72, on the other hand, is clamped between the sliding member 73 and a ring 73a being fitted thereto. The base member 64e of the second valve element 64 is placed on the sliding member 73, and two coil springs 60 and 66 urge those two members 64e and 73 such that they will keep in contact with each other. With the above arrangement, the diaphragm 72 receives differential pressure between discharge

pressure PdH available at one port 41 and discharge pressure PdL at another port 42. This differential pressure displaces the sliding member 73 in its axial direction, causing the second valve element 64 to move toward or away from its corresponding second valve seat 56.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54 and taper-shaped valve element 61 upward as viewed in FIG. 11, making the taper-shaped valve element 61 sit on the first valve seat 45a. The first control valve 30A is fully closed in this state. With the refrigerant at discharge pressure PdH present in the port 41, the maximum differential pressure acts on the diaphragm 72, making the second control valve 30B fully open. The variable displacement compressor 1 thus operates in the minimum capacity condition.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 11. This movement of the plunger 54 allows the taper-shaped valve element 61 to leave the first valve seat 45a and maintain a certain amount of gap between itself and the first valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the hole 64b of the second valve element 64, the central cavity of the cylinder 40c, and the first control valve 30A. Then, in the second control

valve 30B, the diaphragm 72 receives differential pressure between two different discharge pressures PdH and PdL, which moves the sliding member 73 upward as viewed in FIG. 11 so that the differential pressure will become a predetermined level. The second valve element 64 follows this movement of the sliding member 73, thus controlling the refrigerant at discharge pressure PdH that flows through the second control valve 30B, from one port 41 to another port 43.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure acting on the diaphragm 72, the second valve element 64 in the second control valve 30B is impelled in the direction that it gives a greater openness, allowing the second control valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the diaphragm 72 of the second control valve 30B receives a reduced differential pressure and reduces the refrigerant flowing into the crank chamber 15 because its sliding member 73 impels the second valve element 64 in the valve-closing direction. As a result of this control action, the variable displacement compressor 1

operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges. Note here that the second control valve 30B is controlled to be responsive only to differential pressure between two different discharge pressures PdH and PdL, because since its second valve element 64 is decoupled from variations of discharge pressure PdH.

Eleventh Embodiment

FIG. 12 is a sectional view of a capacity control valve for a variable displacement compressor according to an eleventh embodiment of the invention. Since many of the valve components shown in FIG. 12 are identical or similar to those discussed in FIG. 2, 5, or 11, the same reference numerals are used in FIG. 12 to designate such components, and the following section will not provide details about them.

The illustrated capacity control valve 30 of the eleventh embodiment resembles that of the tenth embodiment (FIG. 11) in that both of them use a diaphragm 72 as their pressure sensing element. The eleventh embodiment is, however, different from the tenth embodiment in that the taper-shaped valve element (first valve element) 61 in its first control valve 30A is disposed in an upstream-side space adjacent to a first valve seat 45b formed at the top end of the cylinder 40c. For this reason, in the solenoid unit 30C of the eleventh embodiment, the plunger 54 and

core 53 have swapped their positions on the axis. Also, a shaft 49 is employed to connect the first valve element 61 with the plunger 54 in the solenoid unit 30C. The first valve element 61 is urged by a coil spring 55 in the valve-closing direction.

The eleventh embodiment operates basically in the same way as the tenth embodiment because of its similarity in structure; that is, it uses a diaphragm 72 to detect differential pressure ΔP between the upstream and downstream ends of the first control valve 30A, so as to control the refrigerant flow in the second control valve 30B according to that differential pressure ΔP. In this structure, the widened base member 64e of the second valve element 64 has a round hole 64f in addition to the hole 64b in order to deliver the discharge pressure PdH from the port 41 toward the upstream side of the first valve element 61.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 downward as viewed in FIG. 12, making the first valve element 61 sit on the first valve seat 45b. The first control valve 30A is fully closed in this state. With the refrigerant at discharge pressure PdH present in the port 41, the maximum differential pressure acts on the diaphragm 72, making the second control valve 30B fully open. The variable

displacement compressor 1 thus operates in the minimum capacity condition.

When the solenoid unit 30C is energized, the plunger 54 moves upward as viewed in FIG. 12. This movement of the plunger 54 allows the taper-shaped valve element 61 to leave the first valve seat 45b and maintain a certain amount of gap between itself and the first valve seat 45b. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through round hole 64f and the hole 64b of the second valve element 64, the first control valve 30A, and the central cavity of the cylinder 40c. Then, in the second control valve 30B, the diaphragm 72 receives differential pressure between two different discharge pressures PdH and PdL, which moves the sliding member 73 upward as viewed in FIG. 12 so that the differential pressure will become a predetermined level. The second valve element 64 follows this upward movement of the sliding member 73, thus controlling the flow rate of the refrigerant at discharge pressure PdH that flows through the second control valve 30B, from one port 41 to another port 43.

If the refrigerant flowing through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure acting on the diaphragm 72, the second valve element 64 in the second control valve

30B is moved in the direction that it gives a greater openness, allowing the second control valve 30B to supply more refrigerant into the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the diaphragm 72 of the second control valve 30B detects a reduced differential pressure acting thereon, and thus the sliding member 73 impels the second valve element 64 in the valve-closing direction. As a result, the second control valve 30B reduces the refrigerant supplied to the crank chamber 15, and the variable displacement compressor 1 operates with a larger displacement so as to regulate the flow rate Qd of refrigerant that it discharges.

Twelfth Embodiment

FIG. 13 is a sectional view of a capacity control valve for a variable displacement compressor according to a twelfth embodiment of the invention. Since many of the valve components shown in FIG. 13 are identical or similar to those discussed in FIG. 4, the same reference numerals are used in FIG. 13 to designate such components, and the following section will not provide details about them.

Recall the capacity control valve 30 of the third embodiment shown in FIG. 4. In that embodiment, the first

control valve 30A is located between the discharge chambers 33 and crank chamber 15, and the pressure in the crank chamber 15 is controlled by varying the flow rate of refrigerant at discharge pressure PdL that is supplied from the discharge chambers 33 into the crank chamber 15. Unlike the third embodiment, the twelfth embodiment controls the flow rate of the refrigerant returning from the crank chamber 15 back into the suction chambers 32. In this alternative arrangement, the variable displacement compressor 1 has a fixed orifice in the middle of a passageway that delivers refrigerant from the discharge chambers 33 to the crank chamber 15.

The first control valve 30A and solenoid unit 30C of this capacity control valve 30 have almost the same structure as those in the third embodiment. The exception is that the first control valve 30A is designed to route the discharged refrigerant in the direction that the stream pushes the taper-shaped valve element 61 away from the first valve seat 45a, or in short, in the valve-opening direction.

In the second control valve 30B, there are two pistons 58 and 58a integrally formed with a second valve element 57. The pistons 58 and 58a have the same outer diameter as the valve hole of the second valve seat 56. Discharge pressure PdH acts on the piston 58a and discharge pressure PdL propagates through a communication hole 62 and acts on one endface of the piston 58. Pressure

Pc of the crank chamber 15 is led from the port 43 to an upstream-side cavity adjacent to the second valve element 57. The downstream-side room, on the other hand, communicates with the suction chambers 32 at suction pressure Ps via the port 75. With such an arrangement of the second control valve 30B, the second valve element 57 and piston 58 are responsive to the differential pressure ΔP developed across the first control valve 30A, which is functioning here as an orifice. The second control valve 30B thus controls the flow rate of the refrigerant flowing from the crank chamber 15 to the suction chambers 32 in such a way that the differential pressure ΔP will be maintained at a constant level. This control action varies the capacity of the variable displacement compressor 1 so as to regulate the flow rate of refrigerant being discharged therefrom.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 upward as viewed in FIG. 13, making the first valve element 61 sit on the first valve seat 45a. The first control valve 30A is fully closed in this state.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 13. This movement of the plunger 54 allows the first valve element 61 to leave the first valve seat 45a and maintain a

certain amount of gap between itself and the first valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the second valve element 57 and piston 58, as a single integrated member, receive differential pressure between two different discharge pressures PdH and PdL, in addition to the force of the coil spring 60. The second valve element 57 thus moves to a point at which all those forces and pressures come into balance, which allows the refrigerant in the crank chamber 15 at pressure Pc to flow back to the suction chambers 32. The second control valve 30B can now control the discharge capacity of the variable displacement compressor 1 by varying crank chamber pressure Pc.

The amount of refrigerant flowing through the first control valve 30A may rise due to, for example, sudden acceleration of the engine. If this happens, a larger differential pressure will be produced across that valve 30A. The increased differential pressure actuates the second control valve 30B in the direction that it gives a smaller openness, thus reducing the flow rate of refrigerant coming out of the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A

decreases, the second control valve 30B is actuated in the valve-opening direction, thus increasing the flow rate of refrigerant coming out of the crank chamber 15. As a result of this control action, the variable displacement compressor 1 operates with a larger displacement, thus regulating the flow rate Qd of refrigerant that it discharges.

Thirteenth Embodiment

FIG. 14 is a sectional view of a capacity control valve for a variable displacement compressor according to a thirteenth embodiment of the invention. Since many of the valve components shown in FIG. 14 are identical or similar to those discussed in FIG. 13, the same reference numerals are used in FIG. 14 to designate such components, and the following section will not provide details about them.

The third embodiment (FIG. 4) has presented a capacity control valve 30 that is designed to control the flow rate of refrigerant entering the crank chamber 15, which is referred to as the inflow control. In contrast to this, the twelfth embodiment (FIG. 13) manipulates the flow rate of refrigerant coming out of the crank chamber 15, which is referred to as the outflow control. The thirteenth embodiment now offers a capacity control valve 30 that employs both in-flow and out-flow control mechanisms. More specifically, the capacity control valve

30 of the thirteenth embodiment has a first control valve 30A that is placed on a passageway leading from the discharge chambers 33 and a solenoid unit 30C that governs the cross-sectional area of that passageway. In addition to those components, the capacity control valve 30 has second and third control valves 30B and 30D that detect differential pressure developed across the first control valve 30A and control the pressure in the crank chamber 15 such that the differential pressure will become a specified level.

The second and third control valves 30B and 30D accommodate the following components in their common valve hole: a piston 58, a second valve element 57, and a third valve element 76. Those components are integrally formed as a single member. One edge formed in the valve hole serves as a third valve seat 77, and the piston 58 has the same outer diameter as that valve seat 77. The second valve element 57 receives discharge pressure PdH on its bottom endface, while the piston 58 receives discharge pressure PdL through a communication hole 62. The upstream-side room adjacent to the second valve element 57 is at discharge pressure PdH introduced from a port 41. The downstream side, on the other hand, communicates with the crank chamber 15 through another port 43a, the pressure at which is Pc1. The upstream-side space adjacent to the third valve element 76 receives pressure Pc2 from the crank chamber 15 via yet another port 43b. The

downstream-side space adjacent to the third valve element 76, on the other hand, communicates with the suction chambers 32 at suction pressure Ps via still another port 75.

With the arrangement described above, the piston 58 and second valve element 57 move together in response to differential pressure ΔP across the first control valve 30A, which is functioning here as an orifice. The second and third control valves 30B and 30D now act as a three-way valve that controls the inflow of refrigerant from the discharge chambers 33 into the crank chamber 15, simultaneously with the outflow from the crank chamber 15 to the suction chambers 32, so that the differential pressure ΔP will be maintained at a constant level.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 upward as viewed in FIG. 14, making the first valve element 61 sit on the first valve seat 45a. The first control valve 30A is fully closed in this state.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 14. This movement of the plunger 54 causes the first valve element 61 to leave the first valve seat 45a and maintain a certain amount of gap. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing

out of the port 42 through the first control valve 30A. Then in the second control valve 30B, the unified valve member (i.e., second valve element 57, third valve element 76, and piston 58) receives differential pressure between two different discharge pressures PdH and PdL while being pushed by the coil spring 60, and thus moves to a point at which all those forces and pressures come into balance. The second control valve 30B now supplies the crank chamber 15 with refrigerant at discharge pressure PdH, and the third control valve 30D allows the refrigerant at pressure Pc to flow back into the suction chambers 32. The capacity control valve 30 varies the crank chamber pressure Pc in this way, thus being able to control the discharge capacity of the variable displacement compressor 1.

The amount of refrigerant flowing through the first control valve 30A may rise due to, for example, sudden acceleration of the engine. If this happens, a larger differential pressure will be produced across that valve 30A. The increased differential pressure makes the second control valve 30B open wider, while actuating the third control valve 30D in the valve-closing direction. This control action brings about an increased inflow of refrigerant to the crank chamber 15, along with a decreased outflow from the crank chamber 15. As a result, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original

discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-closing direction, thus producing a decreased inflow of refrigerant to the crank chamber 15. At the same time, the third control valve 30D is impelled in the valve-opening direction, resulting in an increased outflow from the crank chamber 15. The variable displacement compressor 1 now operates with a larger displacement, resulting in a regulated flow rate Qd of refrigerant that it discharges.

Fourteenth Embodiment

FIG. 15 is a sectional view of a capacity control valve for a variable displacement compressor according to a fourteenth embodiment of the invention. Since many of the valve components shown in FIG. 15 are identical or similar to those discussed in FIG. 13, the same reference numerals are used in FIG. 15 to designate such components, and the following section will not provide details about them.

As opposed to the twelfth embodiment (FIG. 13), the capacity control valve 30 of the fourteenth embodiment is designed to control how much of the discharged refrigerant to supply to the crank chamber 15. Another difference is that the second valve element 57 of the second control valve 30B is provided as a discrete component, not integrated with a pressure sensing member

that responds to differential pressure across the first control valve 30A.

More specifically, the second control valve 30B is constructed as follows. A piston 58 is located inside the body 40, and a communication hole 62 is bored through the body 40 to apply discharge pressure PdL to the piston 58. A refrigerant passageway branches from the communication hole 62, leading to a port 43 for the crank chamber 15. In the middle of this passageway, a second valve seat 56 is formed as an integral part of the body 40. Located downstream with respect to the second valve seat 56 is a second valve element 57. This second valve element 57, integrally formed with the piston 58, can move toward and away from the second valve seat 56 in the downstream-side space. The piston 58 receives discharge pressure PdL on its distal end. In addition, a piston 78, coil spring 79, and spring seat 80 are installed coaxially with the second valve element 57 and piston 58 in a space formed between the port 41 and communication hole 62. Discharge pressure PdH is available in this space. A shaft is integrally formed with the second valve element 57, extending therefrom toward the piston 78 in a space that communicates with the communication hole 62. The piston 78 is forced against the shaft by the coil spring 79. Discharge pressure PdL does not affect the movement of the second valve element 57 and piston 58 because their pressure-receiving areas are substantially equal. In the

second control valve 30B, its piston 78 is responsive to differential pressure ΔP developed across the first control valve 30A, which is functioning here as an orifice. The second control valve 30B thus controls the flow rate of refrigerant from the discharge chambers 33 to the crank chamber 15 such that the differential pressure ΔP will be maintained at a constant level. This mechanism varies the capacity of the variable displacement compressor 1 so as to regulate the flow rate of refrigerant that it discharges.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 in the upward direction as viewed in FIG. 15, making the first valve element 61 sit on the first valve seat 45a. The first control valve 30A is fully closed in this state.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 15. This movement of the plunger 54 allows the first valve element 61 to leave the first valve seat 45a and maintain a certain amount of gap between itself and the first valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then in the second control valve 30B, the piston 78 receives differential pressure between two different discharge

pressures PdH and PdL while being pushed by two coil springs 60 and 79, and thus moves to a point at which all those forces and pressures come into balance. This allows the refrigerant at discharge pressure PdH to flow into the crank chamber 15, and the second control valve 30B can now control the discharge capacity of the variable displacement compressor 1 by varying crank chamber pressure Pc.

The amount of refrigerant flowing through the first control valve 30A may rise due to, for example, sudden acceleration of the engine. If this happens, a larger differential pressure will be produced across that valve 30A, which makes the second control valve 30B open wider. This control action produces an increased inflow of refrigerant to the crank chamber 15, and as a result, the variable displacement compressor 1 operates with a smaller displacement, thus recovering its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-closing direction, thus producing a decreased inflow of refrigerant to the crank chamber 15. The variable displacement compressor 1 now operates with a larger displacement, resulting in a regulated flow rate Qd of refrigerant that it discharges.

Fifteenth Embodiment

FIG. 16 is a sectional view of a capacity control

valve for a variable displacement compressor according to a fifteenth embodiment of the invention. Since many of the valve components shown in FIG. 16 are identical or similar to those discussed in FIG. 13, the same reference numerals are used in FIG. 16 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of the fifteenth embodiment is similar to that of the twelfth embodiment (FIG. 13) in that it controls the outflow of refrigerant from the crank chamber 15 to the suction chambers 32. The fifteenth embodiment, however, differs in that the second valve element 57 in its second control valve 30B is provided as a discrete component, not integrated with a pressure sensing member that responds to differential pressure across the first control valve 30A.

More specifically, to detect differential pressure across the first control valve 30A, the second control valve 30B employs a piston 78, a coil spring 79, and a spring seat 80. Ports 43 and 75 are disposed to communicate with the crank chamber 15 and suction chambers 32, respectively, and between these two ports, a second valve seat 56 is formed as an integral part of the body 40. A second valve element 57 is installed in an upstream-side space near the port 43 such that it can move toward and away from the second valve seat 56. Integrally formed with this second valve element 57 is a piston 58 with the same diameter as the valve hole of the second valve seat 56.

Discharge pressure PdL propagates through a communication hole 62 and acts on one endface of the piston 58. The second valve element 57 is integrally formed also with another piston 58a having nearly the same diameter as the valve hole of the second valve seat 56. This piston 58a is held in the body 40 in an airtight manner, movably in its axial direction, receiving discharge pressure PdL. The lower end of the piston 58a as viewed in FIG. 16 abuts on yet another piston 78. Discharge pressure PdL does not affect movement of the pistons 58a and 58 because their diameters are substantially the same. With the above arrangement of the second control valve 30B, the piston 78 is responsive to differential pressure ΔP across the first control valve 30A, which is functioning here as an orifice. The second control valve 30B thus controls the outflow of refrigerant from the crank chamber 15 to the suction chambers 32 in such a way that the differential pressure ΔP will be maintained at a constant level. This control action varies the capacity of the variable displacement compressor 1 so as to regulate the flow rate of refrigerant being discharged therefrom.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 upward as viewed in FIG. 16, making the first valve element 61 sit on the first valve seat 45a. The first control valve 30A

is fully closed in this state.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 16. This movement of the plunger 54 allows the first valve element 61 to leave the first valve seat 45a and maintain a certain amount of gap between itself and the first valve seat 45a. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the piston 78 receives differential pressure between discharge pressures PdH and PdL, while being pushed by two coil springs 60 and 79, and thus moves to a point at which all those forces and pressures come into balance. With this movement of the piston 78, the refrigerant in the crank chamber 15 at pressure Pc is allowed to flow back into the suction chambers 32. The second control valve 30B can now control the discharge capacity of the variable displacement compressor 1 by varying crank chamber pressure Pc.

The amount of refrigerant flowing through the first control valve 30A may rise due to, for example, sudden acceleration of the engine. If this happens, a larger differential pressure will be produced across that valve 30A. The increased differential pressure actuates the second control valve 30B in the direction that it gives a smaller openness, thus reducing the flow rate of refrigerant coming out of the crank chamber 15 and raising

the crank chamber pressure Pc. As a result of this action, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-opening direction, thus permitting an increased outflow from the crank chamber 15. Since the crank chamber pressure Pc goes down, the variable displacement compressor 1 now operates with a larger displacement, resulting in a regulated flow rate Qd of refrigerant that it discharges.

Sixteenth Embodiment

FIG. 17 is a sectional view of a capacity control valve for a variable displacement compressor according to a sixteenth embodiment of the invention. Since many of the valve components shown in FIG. 17 are identical or similar to those discussed in FIGS. 14 and 16, the same reference numerals are used in FIG. 17 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of this sixteenth embodiment is similar to that of the thirteenth embodiment (FIG. 14) in that it controls both inflow and outflow of refrigerant to/from the crank chamber 15. The sixteenth embodiment, however, differs in that the second valve element 57 in its second control valve 30B is provided as

a discrete component, not integrated with a member that senses differential pressure across the first control valve 30A. Regarding the pressure responsive member, the sixteenth embodiment uses a similar structure to that in the fifteenth embodiment (FIG. 16).

More specifically, inside the second and third control valves 30B and 30D, a piston 58, a second valve element 57, and a third valve element 76 are disposed in an integrated manner. The piston 58 has the same outer diameter as the valve holes of second and third valve seats 56 and 77 so as to avoid the effect of discharge pressure PdL acting thereon. With this arrangement, the piston 58 and second valve element 57 move together in response to differential pressure ΔP across the first control valve 30A, which is functioning here as an orifice. The second and third control valves 30B and 30D now serve as a three-way valve that controls the inflow of refrigerant from the discharge chambers 33 into the crank chamber 15, simultaneously with the outflow from the crank chamber 15 to the suction chambers 32, in such a way that the differential pressure ΔP will be maintained at a constant level.

The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is in de-energized state, the coil spring 55 urges the plunger 54, shaft 49, and first valve element 61 upward as viewed in FIG. 17, making the first valve element 61 sit

on the first valve seat 45a. The first control valve 30A is fully closed in this state.

When the solenoid unit 30C is energized, the plunger 54 moves downward as viewed in FIG. 17. This movement of the plunger 54 allows the first valve element 61 to leave the first valve seat 45a and maintain a certain amount of gap. As a result, the refrigerant coming into the port 41 at discharge pressure PdH begins flowing out of the port 42 through the first control valve 30A. Then in the second control valve 30B, the unified valve member (i.e., second valve element 57, third valve element 76, and piston 58) receives differential pressure between discharge pressures PdH and PdL while being pushed by the coil springs 60 and 79, and thus moves to a point at which all those forces and pressures come into balance. The second control valve 30B now supplies refrigerant at pressure Pc1 to the crank chamber 15 by controlling refrigerant at discharge pressure PdL, and at the same time, the third control valve 30D allows the refrigerant at pressure Pc2 in the crank chamber 15 to flow back into the suction chambers 32. The capacity control valve 30 varies the crank chamber pressure Pc in this way, thus being able to control the discharge capacity of the variable displacement compressor 1.

The amount of refrigerant flowing through the first control valve 30A may rise due to, for example, sudden acceleration of the engine. If this happens, a

larger differential pressure will be produced across that valve 30A. The increased differential pressure makes the second control valve 30B open wider, while actuating the third control valve 30D in the valve-closing direction. This control action produces an increased inflow of refrigerant to the crank chamber 15, together with a decreased outflow from the crank chamber 15. As a result, the variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If, in turn, the refrigerant flowing through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-closing direction, and the third control valve 30D in the valve-opening direction, thus producing a decreased inflow of refrigerant to the crank chamber 15 and an increased outflow from the crank chamber 15. The variable displacement compressor 1 now operates with a larger displacement, resulting in a regulated flow rate Qd of refrigerant that it discharges.

Seventeenth Embodiment

FIG. 18 is a sectional view of a capacity control valve for a variable displacement compressor according to a seventeenth embodiment of the invention. Since many of the valve components shown in FIG. 18 are identical or similar to those discussed in FIG. 15, the same reference numerals are used in FIG. 18 to designate such components,

and the following section will not provide details about them.

As in the fourteenth embodiment (FIG. 15), the capacity control valve 30 of this seventeenth embodiment is designed to control how much of the discharged refrigerant to supply to the crank chamber 15. Another similarity is that the second valve element 57 of the second control valve 30B is provided as a discrete component, not integrated with a member that responds to differential pressure across the first control valve 30A. The seventeenth embodiments is, however, different in that it has no back-pressure cancellation mechanism for the second valve element 57.

More specifically, the second control valve 30B is constructed as follows. A second valve element 57 is urged by a coil spring 60 in the valve-closing direction, where discharge pressure PdL is routed through a communication hole 62 and acts only on the piston 78 and second valve element 57. The upper end of the coil spring 60, as viewed in FIG. 18, is supported by a lid 59c having a vent. An O-ring 29b is used for sealing of the capacity control valve 30 when it is installed in a variable displacement compressor 1. The upper space above the level of this O-ring 29b, as viewed in FIG. 18, will be at pressure Pc, i.e., the pressure in the port 43, meaning that the same pressure Pc will be available in the cavity where the coil spring 60 resides.

The above-described capacity control valve 30 bears close resemblance to the fourteenth embodiment (FIG. 15) in terms of the structure, except for the fact that the second valve element 57 is not free from back pressures. When its solenoid unit 30C is in de-energized state, the capacity control valve 30 operates in the same way as described in the fourteenth embodiment. This similarity in its control operations also applies when the solenoid unit 30C is energized, as well as when the engine rotation varies.

Eighteenth Embodiment

FIG. 19 is a sectional view of a capacity control valve for a variable displacement compressor according to an eighteenth embodiment of the invention. Since many of the valve components shown in FIG. 19 are identical or similar to those discussed in FIG. 16, the same reference numerals are used in FIG. 19 to designate such components, and the following section will not provide details about them.

As in the fifteenth embodiment (FIG. 16), the capacity control valve 30 of this eighteenth embodiment is designed to control the outflow of refrigerant coming out of the crank chamber 15 to the suction chambers 32. Another likeness is that the second valve element 57 of the second control valve 30B is provided as a discrete component, not integrated with a member that senses

differential pressure across the first control valve 30A. The eighteenth embodiments is, however, different in that it has no back-pressure cancellation mechanism for the second valve element 57.

More specifically, the second control valve 30B is constructed as follows. A second valve element 57 is urged against a piston 78 by a coil spring 60 in the valve-opening direction, where discharge pressure PdL is routed through a communication hole 62 in such a way that it acts only on the piston 78 and another piston that extends from the second valve element 57. Yet another piston 58 is integrally formed with the second valve element 57, and the coil spring 60 is accommodated in a space between this piston 58 and a lid 59c having a vent. The coil spring space is pressurized at Ps through the vent in the lid 59c.

The above-described capacity control valve 30 bears close resemblance to the fifteenth embodiment (FIG. 16) in terms of the structure, except for the fact that the second valve element 57 is not free from back pressures. When its solenoid unit 30C is in de-energized state, the capacity control valve 30 operates in the same way as described in the fifteenth embodiment. This similarity in its control operations also applies when the solenoid unit 30C is energized, as well as when the engine rotation varies.

Nineteenth Embodiment

FIG. 20 is a sectional view of a capacity control valve for a variable displacement compressor according to a nineteenth embodiment of the invention. Since many of the valve components shown in FIG. 20 are identical or similar to those discussed in FIG. 17, the same reference numerals are used in FIG. 20 to designate such components, and the following section will not provide details about them.

The capacity control valve 30 of this nineteenth embodiment controls both inflow and outflow of refrigerant to/from the crank chamber 15, as in the seventeenth embodiment (FIG. 18). The nineteenth embodiment is also similar to the seventeenth embodiment in that the second valve element 57 in its second control valve 30B is provided as a discrete component, not integrated with a member that senses differential pressure across the first control valve 30A.

More specifically, the second and third control valves 30B and 30D are constructed as follows. A second valve element 57 and a third valve element 76, which constitute a three-way valve, are urged by a coil spring 60 in the valve-closing direction and in the valve-opening direction, respectively, where discharge pressure PdL is routed through a communication hole 62 in such a way that it acts only on the second valve element 57 and piston 78. A piston 58 is integrally formed with the second and third valve elements 57 and 76, and the coil spring 60 is

accommodated in a space between this piston 58 and a lid 59c having a vent. The coil spring space is pressurized at Ps through the vent in the lid 59c.

The above-described capacity control valve 30 bears close resemblance to the sixteenth embodiment (FIG. 17) in terms of the structure, except for the fact that the second valve element 57 and third valve element 76 are not free from back pressures. When its solenoid unit 30C is in de-energized state, the capacity control valve 30 operates in the same way as described in the sixteenth embodiment. This similarity in its control operations also applies when the solenoid unit 30C is energized, as well as when the engine rotation varies.

Various types of capacity control valves 30 have been presented as preferred embodiments of the present invention. All those embodiments share a common concept that the first control valve 30A controls the cross-section area of a passageway of discharged refrigerant, and the second control valve 30B (and third control valve 30D in several cases) controls pressure Pc in the crank chamber 15 in such a way that the differential pressure across the controlled passageway will be maintained at a specified level. The capacity control valves of the present invention should not be limited to the structure that uses differential pressure on the discharge side of the valves. Rather, it has to be appreciated that the invention covers the structure using differential pressure

on the suction side. That is, the first control valve 30A may control the cross-section area of a passageway of refrigerant coming into the compressor, and the second control valve 30B (and third control valve 30D) controls pressure Pc in the crank chamber 15 in such a way that the differential pressure across the controlled passageway will be maintained at a specified level.

As can be seen from the above explanation, the present invention proposes a structure that employs first and second control valves formed in an integrated way. Here, the first control valve controls the cross-section area at a midway point between low-pressure refrigerant passage and suction chamber, or between discharge chamber and high-pressure refrigerant passage, according to a given external condition. The second control valve, on the other hand, detects differential pressure between upstream end and downstream end of the first control valve and controls the crank chamber pressure in such a way that the differential pressure will be maintained at a specified level. This feature of the present invention enables us to construct a smaller variable displacement compressor at lower cost.

Because the first control valve has only to produce a small amount of differential pressure, the solenoid unit can drive the valve with a small power, and thus it is not necessary to increase its size to achieve the purpose. The present invention can easily be applied

to refrigeration cycles using HFC-134a in a system that should operate with small differential pressure between discharge chamber and crank chamber, or crank chamber and suction chamber. In addition, the present invention can also be applied to those using high-pressure refrigerant in its supercritical region.

The foregoing is considered as illustrative only of the principles of the present invention. Further, since numerous modifications and changes will readily occur to those skilled in the art, it is not desired to limit the invention to the exact construction and applications shown and described, and accordingly, all suitable modifications and equivalents may be regarded as falling within the scope of the invention in the appended claims and their equivalents.